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Nishida

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(54) **HYDRAULIC PRESSURE CONTROLLER FOR CONTINUOUSLY VARIABLE TRANSMISSION**

(75) Inventor: **Naofumi Nishida**, Toyota (JP)

(73) Assignee: **TOYOTA JIDOSHA KABUSHIKI KAISHA**, Toyota-shi (JP)

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(52) **U.S. Cl.**
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(58) **Field of Classification Search**
CPC F16H 61/6625; F16H 2061/66286
See application file for complete search history.

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Primary Examiner — F. Daniel Lopez
(74) *Attorney, Agent, or Firm* — Oblon, McClelland, Maier & Neustadt, L.L.P.

(57) **ABSTRACT**

Disclosed is a hydraulic control device for a continuously variable transmission which changes the transmission gear ratio by changing the hydraulic pressure supplied to hydraulic pressure chambers for pulleys in the continuously variable transmission via control valves is changed. In this hydraulic control device, line pressure regulation hydraulic pressure which corresponds to the magnitudes of the oil pressure supplied to the hydraulic pressure chambers is introduced into a regulator valve, and line pressure which is input into the control valves according to the magnitudes of the oil pressure is subjected to feedback regulation. In the hydraulic control device for the continuously variable transmission, after the valve element of the control valve is driven in the valve opening direction, a vibration restriction control operation which temporarily restricts the displacement of the valve element is executed.

7 Claims, 6 Drawing Sheets

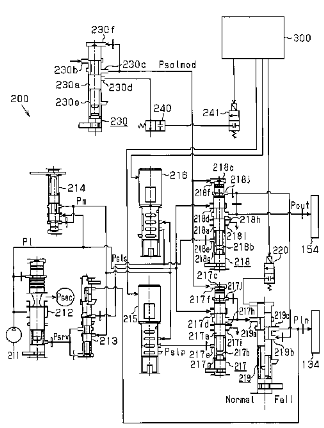
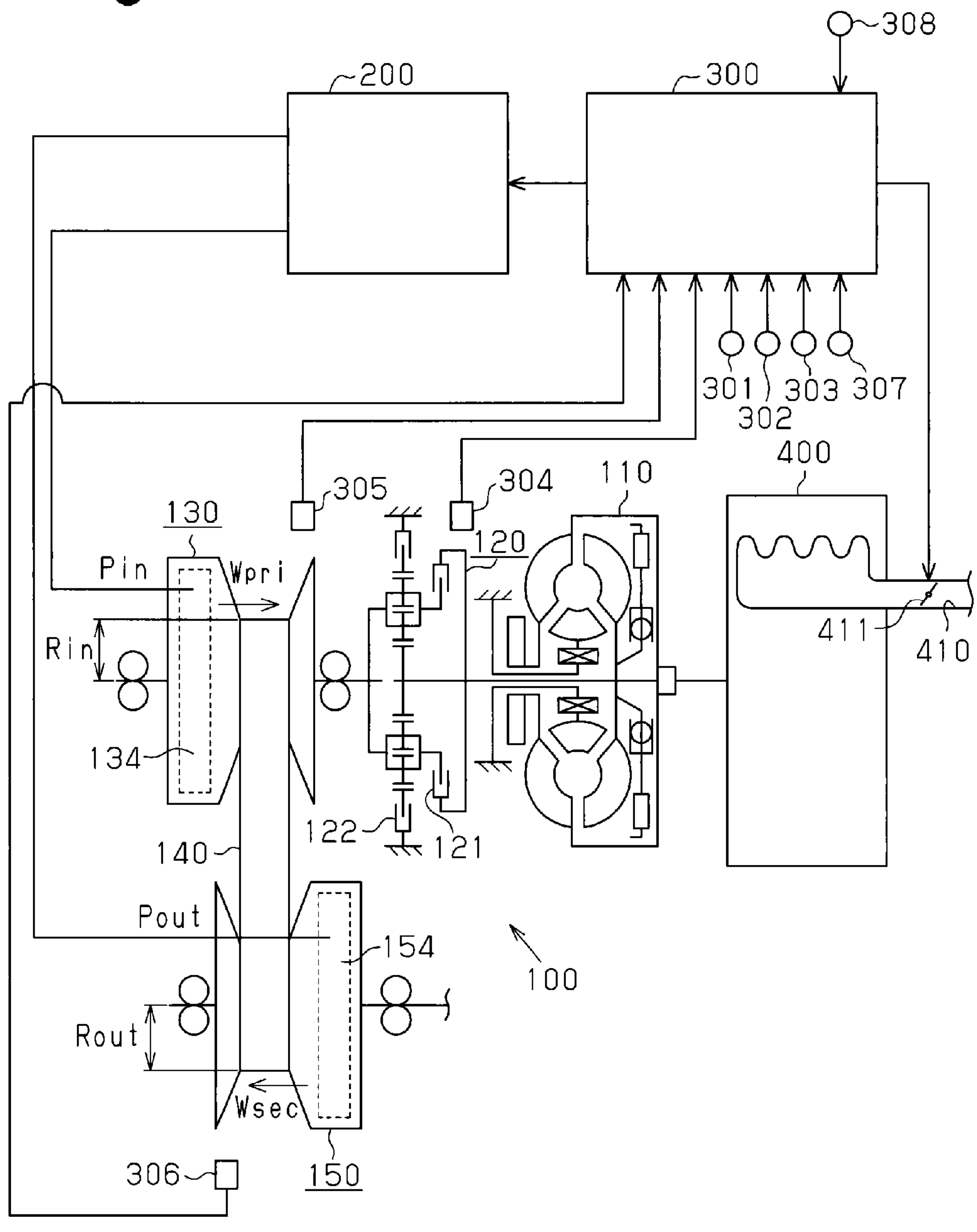
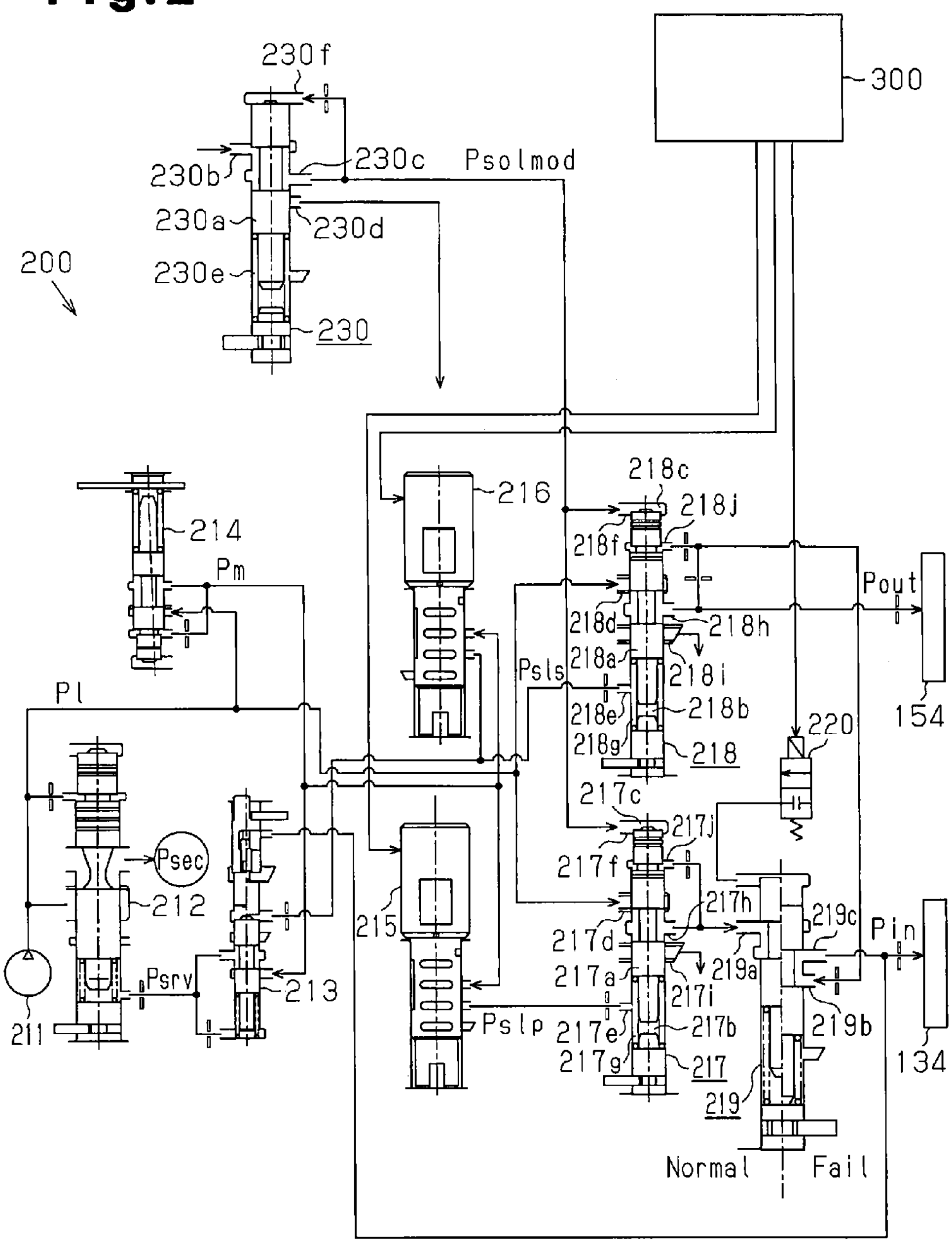


Fig.1



308
200
300
301
302
303
307
400
305
304
110
130
120
Pin
Wpri
Rin
411
410
134
122
121
140
Pout
154
100
Rout
Wsec
306
150

Fig.2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218h
218a
218i
Psls
218e
218b
Pl
218g
218
154
220
217c
217j
Psec
217f
212
217h
219c
215
217d
Pin
217a
219a
Psrv
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217i
219b
Pslp
217e
217b
211
217g
217
134
219
Normal
Fail

Fig.3

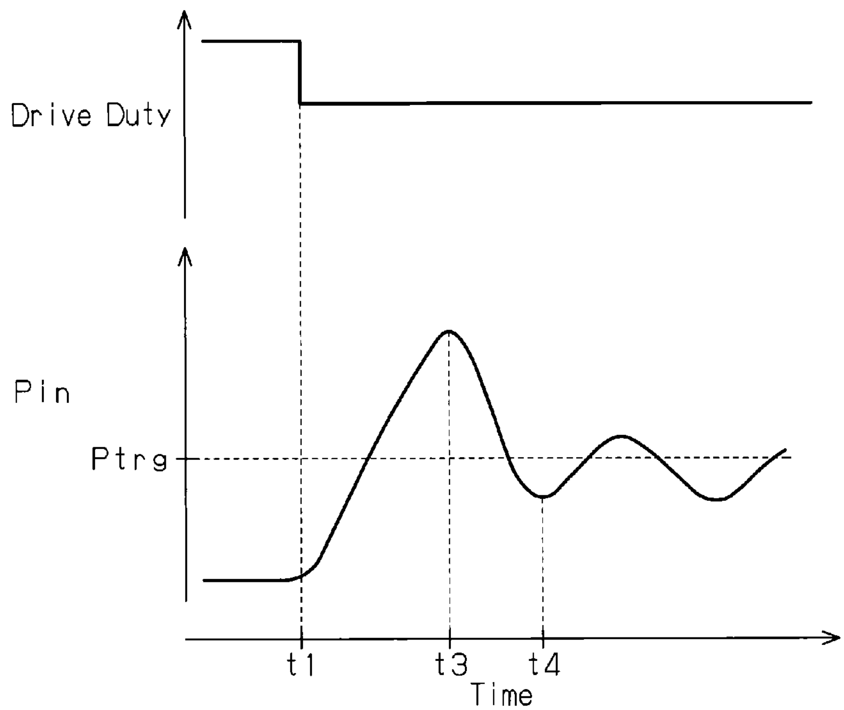

Drive Duty
Pin
Ptrg
t1
t3
t4
Time

Fig.4

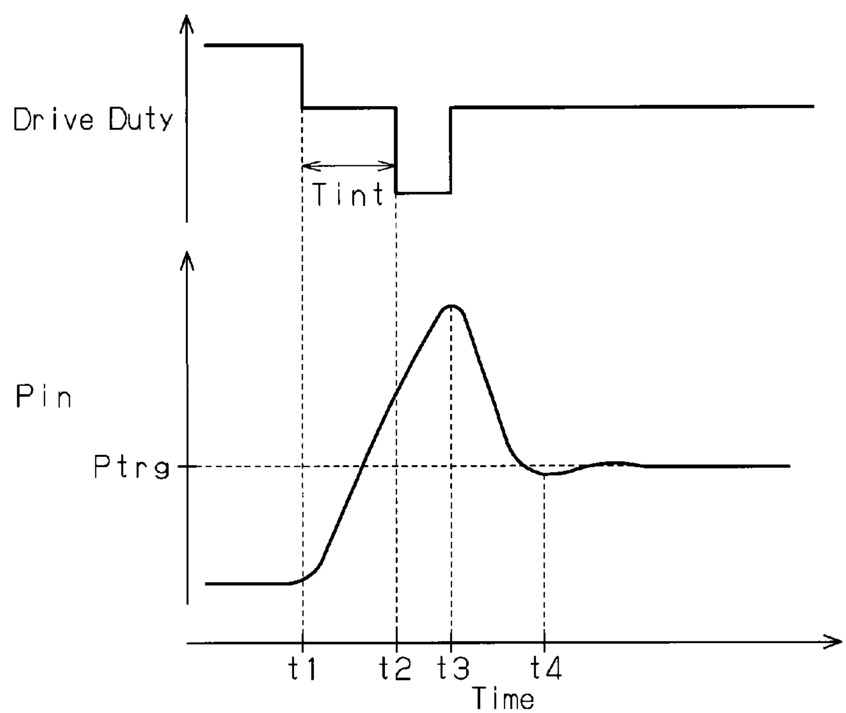

Drive Duty
Tint
Pin
Ptrg
t1
t2
t3
t4
Time

Fig.5

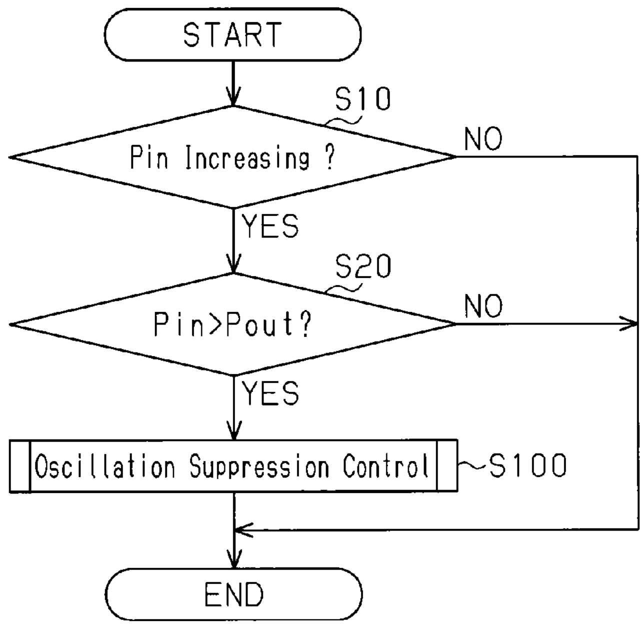

START
S10
Pin Increasing ?
NO
YES
S20
Pin>Pout?
NO
YES
Oscillation Suppression Control
S100
END

Fig.6

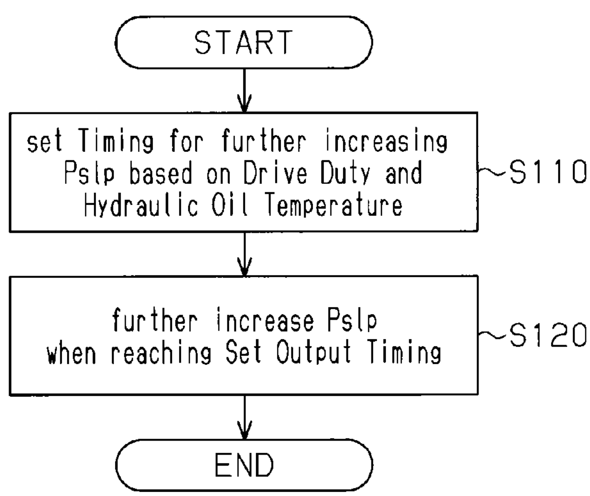
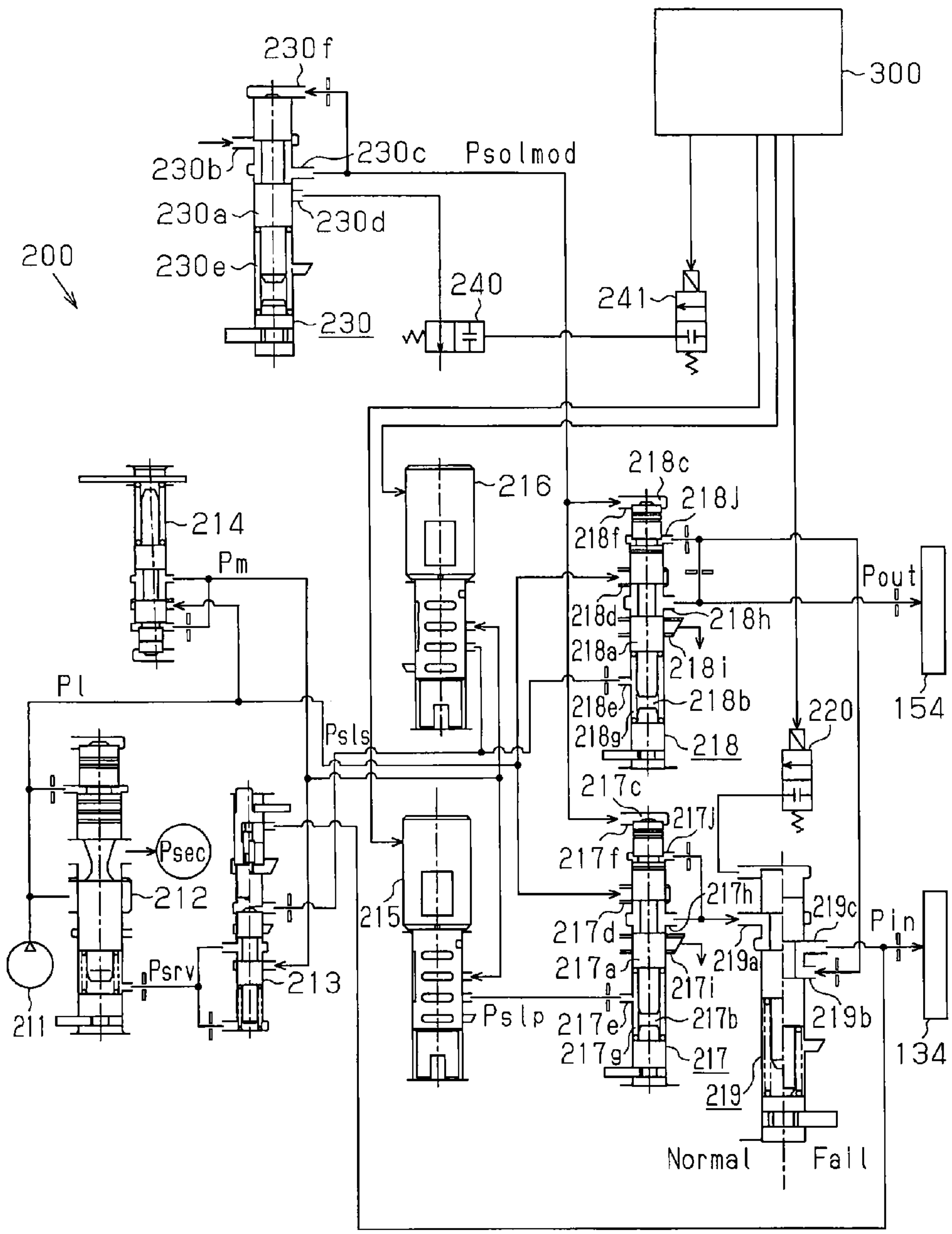

START
set Timing for further increasing Pslp based on Drive Duty and Hydraulic Oil Temperature
S110
further increase Pslp when reaching Set Output Timing
S120
END

Fig.7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218b
154
220
Psls
218g
218
217c
217J
217f
Psec
217h
219c
212
215
217d
Pin
219a
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213
217e
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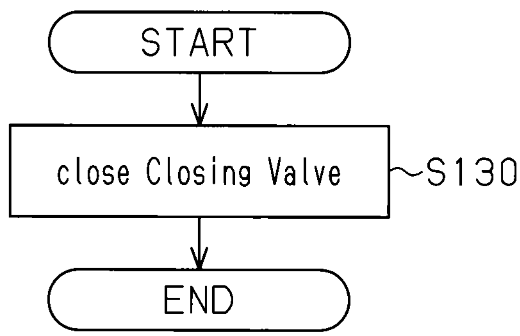
217
134
219
Normal
Fail

Fig.8



HYDRAULIC PRESSURE CONTROLLER FOR CONTINUOUSLY VARIABLE TRANSMISSION

FIELD OF DISCLOSURE

The present invention relates to hydraulic pressure controllers for a continuously variable transmission that controls the hydraulic pressure supplied to each pulley of a belt type continuously variable transmission, and in particular, to a hydraulic pressure controller for a continuously variable transmission that feedback-adjusts a line pressure, which is an originating pressure of the hydraulic pressure supplied to each pulley, in accordance with the level of the hydraulic pressure supplied to each pulley.

BACKGROUND OF THE DISCLOSURE

A belt type continuously variable transmission is a known transmission mounted on a vehicle or the like. The belt type continuously variable transmission includes a first pulley that receives driving force from an internal combustion engine, a second pulley coupled to a vehicle wheel, and a belt running around the two pulleys. The continuously variable transmission changes the winding radius of the belt at each pulley to change the gear ratio in a continuous and stepless manner.

Such a belt type continuously variable transmission changes the hydraulic pressure of a hydraulic pressure chamber arranged in each pulley to change the balance of thrusts, which is the force that sandwiches the belt. This changes the winding ratio of the belt at each pulley and controls the gear ratio.

To this end, such a continuously variable transmission includes a hydraulic pressure controller that controls the hydraulic pressure supplied to each pulley. The hydraulic pressure controller includes a plurality of solenoid valves, which are driven based on an electrical drive command, and a plurality of control valves, which are driven by a driving hydraulic pressure output from the solenoid valves. The hydraulic pressure controller drives the control valves based on drive commands output from an electronic control unit to supply hydraulic oil to the hydraulic pressure chamber of each pulley or discharge hydraulic oil from the hydraulic pressure chamber of each pulley thereby controlling the hydraulic pressure of the hydraulic pressure chamber in each pulley.

In the hydraulic pressure controller for the continuously variable transmission, a regulator valve adjusts the hydraulic pressure of the hydraulic oil discharged from an oil pump and generates line pressure, which is the originating pressure of the hydraulic pressure to output from the control valves. The line pressure is input to the control valves, and the line pressure is adjusted by the control valves to generate the hydraulic pressure supplied to each pulley.

In the hydraulic pressure controller described above, a drive load of the oil pump becomes high when the line pressure becomes high. Thus, a hydraulic pressure controller for a continuously variable transmission that is described in patent document 1 feedback-adjusts the line pressure in accordance with the level of the hydraulic pressure supplied to each pulley.

Specifically, hydraulic pressure corresponding to the level of the hydraulic pressure supplied to each pulley is input to the regulator valve. The line pressure is increased when the hydraulic pressure supplied to each pulley is high, and the line pressure is decreased when the hydraulic pressure supplied to each pulley is low.

In this manner, by feedback adjusting the line pressure in accordance with the level of the hydraulic pressure supplied to each pulley, an excessive increase in the drive load of the oil pump when the line pressure increases more than necessary is suppressed.

PRIOR ART DOCUMENT

Patent Documents

Patent document 1: Japanese Laid-Open Patent Publication No. 2009-156413

DISCLOSURE OF THE INVENTION

Problems that are to be Solved by the Invention

When changing the hydraulic pressure supplied to each pulley, valve bodies of the control valves that control the hydraulic pressure supplied to the pulleys are driven, and a valve body of the regulator valve adjusts the line pressure. However, immediately after starting the driving of the valve bodies, inertia may excessively move the valve bodies and thereby oscillate the valve bodies.

If a valve body oscillates, the hydraulic pressure adjusted by the control valves and the regulator valve oscillates over a target hydraulic pressure. When the hydraulic pressure supplied to each pulley oscillates, the tension on the belt repetitively increases and decreases. This may result in slipping of the belt on each pulley or excessive load being applied to the belt and thereby lower the durability of the continuously variable transmission.

Further, once the hydraulic pressure supplied to each pulley starts to oscillate, the oscillating hydraulic pressure is fed back to the regulator valve and the line pressure is adjusted based on the fed back hydraulic pressure. As a result, the line pressure adjusted through the regulator valve is also oscillated. This may produce an adverse cycle in which the line pressure is feedback adjusted based on the oscillating hydraulic pressure, the oscillation is propagated to the line pressure, and the hydraulic pressure supplied to each pulley is adjusted based on the oscillating line pressure. As a result, a state in which the hydraulic pressure supply to each pulley oscillates may continue over a long period of time.

It is an object of the present invention to provide a hydraulic pressure controller for a continuously variable transmission capable of feedback adjusting the line pressure in accordance with the level of the hydraulic pressure supplied to each pulley and suppressing oscillation of the hydraulic pressure supplied to each pulley and the line pressure when changing the hydraulic pressure supplied to each pulley, while suppressing the drive load of the oil pump from becoming excessively large.

Means for Solving the Problems

To achieve the above object, after driving a valve body of the control valve in a valve opening direction, a hydraulic pressure controller according to the present invention executes oscillation suppression control that temporarily suppresses movement of the valve body.

Thus, oscillation of the valve body when the valve body is driven in the valve opening direction can be suppressed, and repetitive increase and decrease of the hydraulic pressure supplied to a pulley over a target hydraulic pressure is suppressed. Further, by suppressing such oscillation of the valve body and suppressing the oscillation of the hydraulic pressure supplied to the pulley, oscillation of the line pressure, which is feedback adjusted based on the hydraulic pressure supplied to the pulley, can be suppressed.

This can suppress the occurrence of an adverse cycle in which the line pressure is adjusted based on the oscillating hydraulic pressure and the hydraulic pressure supplied to each pulley is adjusted based on the oscillating line pressure.

In this manner, a hydraulic pressure controller for a continuously variable transmission according to the present invention feedback adjusts the line pressure in accordance with the level of the hydraulic pressure supplied to each pulley, suppresses excessive increases in the drive load applied to an oil pump, and suppresses oscillation of the hydraulic pressure, which is supplied to each pulley, and the line pressure when the hydraulic pressure supplied to each pulley is changed. This consequently suppresses repetitive increasing and decreasing of tension on a belt when the hydraulic pressure of the pulleys is changed and suppresses a decrease in the durability of the continuously variable transmission.

When the control valve uses a driving hydraulic pressure input to the control valve to drive the valve body in a valve opening direction, specifically, it is preferred that after starting to output the driving hydraulic pressure to the control valve to drive the valve body in the valve opening direction, oscillation suppression control is executed to further change the driving hydraulic pressure to suppress movement of the valve body in a valve closing direction when the valve body oscillates as the output of the driving hydraulic pressure starts.

More specifically, when the control valve moves the valve body in the valve opening direction as the driving hydraulic pressure increases, after starting the output of the driving hydraulic pressure to drive the valve body in the valve opening direction, the oscillation suppression control further increases the driving hydraulic pressure in accordance with a timing at which the valve body starts to move in the valve closing direction. The employment of such a configuration further increases the driving hydraulic pressure and offsets force that acts to move the valve body in the valve opening direction, suppresses movement of the valve body in the valve closing direction, and suppresses oscillation of the valve body.

After starting the output of the driving hydraulic pressure to the control valve to drive the valve body in the valve opening direction, the timing at which the valve body starts to move in the valve closing direction changes in accordance with the responsiveness or the like of the actual movement of the valve body relative to changes in the driving hydraulic pressure. When the temperature of the hydraulic oil is high, the viscosity of the hydraulic oil decreases. Thus, the responsiveness of the actual movement of the valve body relative to changes in the driving hydraulic pressure increases as the temperature of the hydraulic oil changes. That is, the valve body moves more readily when the driving hydraulic pressure is output as the temperature of the hydraulic oil increases. Further, after the driving hydraulic pressure is output, the timing at which the valve body starts to move in the valve closing direction is advanced as the temperature of the hydraulic oil increases.

Thus, after starting the output of the driving hydraulic pressure to the control valve to drive the valve body in the valve opening direction, it is preferred that a timing for further changing the driving hydraulic pressure be advanced as the temperature of the hydraulic oil increases. The employment of such a configuration allows for the timing for changing the hydraulic pressure to be set in accordance with changes in the responsiveness of the actual movement of the valve body relative to changes in the driving hydraulic pressure as the temperature of the hydraulic oil changes.

Further, as the hydraulic pressure output to the control valve to drive the valve body in the valve opening direction increases, the valve opening speed of the valve body when the driving hydraulic pressure is output increases. That is, as the hydraulic pressure output to drive the valve body in the valve opening direction increases, the valve body moves more readily when the valve body starts to move in the valve closing direction. Thus, as the hydraulic pressure output to the control valve to drive the valve body in the valve opening direction increases, after starting the output of the driving hydraulic pressure to the control valve to drive the valve body in the valve opening direction, the timing for further changing the driving hydraulic pressure may be advanced. When employing such a configuration, the timing for changing the driving hydraulic pressure may be set in accordance with changes in the valve opening speed of the valve body when the driving hydraulic pressure is output.

When the hydraulic pressure controller for a continuously variable transmission includes a control valve provided with a first pressure chamber and a second pressure chamber located on opposite sides of the valve body, and the control valve changes a level of the driving hydraulic pressure supplied to the first pressure chamber to drive the valve body, the movement of the valve body may be suppressed by prohibiting discharge of the hydraulic oil from the second pressure chamber and supply of the hydraulic oil to the second pressure chamber.

The oscillation of the valve body is apt to occur when the driving hydraulic oil acts on the valve body to drive the valve body in the valve opening direction.

Thus, when the hydraulic pressure controller for a continuously variable transmission feedback adjusts the line pressure in correspondence with a change in a larger one of a hydraulic pressure supplied to a first pulley, which is coupled to an internal combustion engine, and a hydraulic pressure supplied to a second pulley, which is coupled to a vehicle wheel, it is preferred that the execution conditions for executing oscillation suppression control be set so that when increasing the hydraulic pressure supplied to the first pulley, the oscillation suppression control is executed when the hydraulic pressure supplied to the first pulley becomes greater than the hydraulic pressure supplied to the second pulley.

The employment of such a configuration executes the oscillation suppression control under a situation that may result in an adverse cycle in which the hydraulic pressure supplied to each pulley oscillates continuously over a long period of time such as when the line pressure is feedback adjusted in correspondence with changes in the hydraulic pressure supplied to the first pulley when increasing the hydraulic pressure supplied to the first pulley.

Thus, a state in which the hydraulic pressure supplied to each pulley oscillates continuously over a long period of time is suppressed in a preferred manner.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic diagram showing a continuously variable transmission that is a control subject of a hydraulic pressure controller according to a first embodiment of the present invention.

FIG. **2** is a schematic diagram showing the structure of a hydraulic pressure control unit in the hydraulic pressure controller of the first embodiment.

FIG. **3** is a time chart showing the relationship of a change in the drive duty of a first solenoid valve and a change in the hydraulic pressure supplied to a first pulley in a gear shift control of the prior art.

FIG. **4** is a time chart showing the relationship of a change in the drive duty of a first solenoid valve and a change in the hydraulic pressure supplied to the first pulley when a oscillation suppression control of the first embodiment is executed.

FIG. **5** is a flowchart showing a flow of a series of processes related to the execution of the oscillation suppression control of the first embodiment.

FIG. **6** is a flowchart showing a flow of processes in the oscillation suppression control of the first embodiment.

FIG. **7** is a schematic diagram showing the structure of a hydraulic pressure control unit in a hydraulic pressure controller according to a second embodiment.

FIG. **8** is a flowchart showing a flow of processes in the oscillation suppression control of the second embodiment.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

First Embodiment

One embodiment of a hydraulic pressure controller for a continuously variable transmission according to the present invention applied to an electronic control unit **300** that controls a continuously variable transmission **100**, which is installed in a vehicle, and a hydraulic pressure control unit **200** will now be described with reference to FIGS. **1** to **6**. FIG. **1** is a schematic diagram showing the structure of the continuously variable transmission **100**, which is a control subject of the hydraulic pressure controller of the present invention.

As shown in FIG. **1**, an input shaft of a torque converter **110** in the continuously variable transmission **100** is connected to an output shaft of an internal combustion engine **400**. An output shaft of the torque converter **110** is connected to an input shaft of a switching mechanism **120**.

The switching mechanism **120** is a double-pinion planet gear mechanism and includes a forward clutch **121** and a reverse brake **122**. An output shaft of the switching mechanism **120** is connected to a first pulley **130**.

When the forward clutch **121** is engaged and the reverse brake **122** is released, the driving force of the internal combustion engine **400**, which is input through the torque converter **110**, is directly transmitted to the first pulley **130**. In contrast, when the forward clutch **121** is released and the reverse brake **122** is engaged, the driving force of the internal combustion engine **400**, which is input through the torque converter **110**, is reversed and transmitted to the first pulley **130** as a driving force of a reversed rotation.

In the switching mechanism **120**, if the forward clutch **121** and the reverse brake **122** are both released, the transmission of the driving force between the internal combustion engine **400** and the first pulley **130** is cut off.

The first pulley **130**, which is coupled to the internal combustion engine **400** by the torque converter **110** and the switching mechanism **120**, is coupled to a second pulley **150**, which is an output side pulley, by a belt **140**. More specifically, a single belt **140** runs around the first pulley **130** and the second pulley **150** arranged in parallel as shown at the lower left side in FIG. **1**. Driving force is transmitted between the first pulley **130** and the second pulley **150** by the belt **140**.

The second pulley **150** is coupled to a differential through a reduction gear (not shown). Thus, the rotation of the second pulley **150** is transmitted to the differential via the reduction gear, and the rotation is transmitted to left and right drive wheels via the differential.

The first pulley **130** is formed by combining a fixed sheave and a movable sheave. A hydraulic pressure chamber **134** is defined and formed in the first pulley **130** as shown by broken lines in FIG. **1**.

The second pulley **150** is also formed by combining a fixed sheave and a movable sheave. A hydraulic pressure chamber **154** is also defined and formed in the second pulley **150** as shown by broken lines in FIG. **1**.

The belt **140** runs around the first pulley **130** and the second pulley **150**, as described above. The belt **140** is sandwiched between the fixed sheave and the movable sheave of the first pulley **130** and sandwiched between the fixed sheave and the movable sheave of the second pulley **150**.

Thus, when a hydraulic pressure Pin of the hydraulic pressure chamber **134** in the first pulley **130** is changed, the distance between the fixed sheave and the movable sheave of the first pulley **130** changes, and thrust Wpri acting on the belt **140** at the first pulley **130** changes. Further, when a hydraulic pressure Pout of the hydraulic pressure chamber **154** in the second pulley **150** changes, the distance between the fixed sheave and the movable sheave of the second pulley **150** changes, and thrust Wsec acting on the belt **140** at the second pulley **150** changes.

As shown in FIG. **1**, each of the pulleys **130** and **150** includes a gradient at a portion that contacts the belt **140**. Thus, when changing the thrust Wpri at the first pulley **130** and the thrust Wsec at the second pulley **150**, winding radii Rin and Rout of the belt **140** at the pulleys **130** and **150** are changed.

More specifically, when increasing the hydraulic pressure Pin in the first pulley **130** to increase the thrust Wpri and decreasing the hydraulic pressure Pout in the second pulley **150** to decrease the thrust Wsec, the winding radius Rin of the belt **140** at the first pulley **130** is increased and the winding radius Rout of the belt **140** at the second pulley **150** is decreased. When decreasing the hydraulic pressure Pin in the first pulley **130** to decrease the thrust Wpri and increasing the hydraulic pressure Pout in the second pulley **150** to increase the thrust Wsec, the winding radius Rin of the belt **140** at the first pulley **130** is decreased and the winding radius Rout of the belt **140** at the second pulley **150** is increased.

In the continuously variable transmission **100**, the hydraulic pressures Pin and Pout of the pulleys **130** and **150** are changed to change the thrusts Wpri and Wsec and change the winding radii Rin and Rout of the belt **140** at the pulleys **130** and **15**. This controls a gear ratio γ.

More specifically, when performing upshifting and decreasing the gear ratio γ, the hydraulic pressure Pin of the hydraulic pressure chamber **134** in the first pulley **130** is increased to increase the thrust Wpri at the first pulley **130**, and the hydraulic pressure Pout of the hydraulic pressure chamber **154** in the second pulley **150** is decreased to decrease the thrust Wsec at the second pulley **150**. This increases the winding radius Rin of the belt **140** at the first pulley **130**, decreases the winding radius Rout of the belt **140** at the second pulley **150**, and decreases the gear ratio γ.

When performing downshifting and increasing the gear ratio γ, the hydraulic pressure Pin of the hydraulic pressure chamber **134** in the first pulley **130** is decreased to decrease the thrust Wpri at the first pulley **130**, and the hydraulic pressure Pout of the hydraulic pressure chamber **154** in the second pulley **150** is increased to increase the thrust Wsec at the second pulley **150**. This decreases the winding radius Rin of the belt **140** at the first pulley **130**, increases the winding radius Rout of the belt **140** at the second pulley **150**, and increases the gear ratio γ.

As shown in FIG. **1**, the hydraulic pressure chambers **134** and **154** of the pulleys **130** and **150** are connected to the hydraulic pressure control unit **200**. The hydraulic pressure control unit **200** is a hydraulic pressure circuit including a plurality of solenoid valves, which are driven based on a drive command output from the electronic control unit **300**, and a control valve, which is driven by the drive hydraulic pressure output from the solenoid valves. The hydraulic pressure of the hydraulic oil is adjusted by operating the control valve to supply the hydraulic oil to the hydraulic pressure chambers **134** and **154** or discharge the hydraulic oil from the hydraulic pressure chambers **134** and **154** thereby adjusting the hydraulic pressures Pin and Pout of the hydraulic pressure chambers **134** and **154**.

The electronic control unit **300** includes a central processing unit (CPU) that executes calculations related to the control of the internal combustion engine **400**, calculations related to the control of the continuously variable transmission **100** through the hydraulic pressure control unit **200**, and the like. Further, the electronic control unit **300** includes a read-out only memory (ROM), which stores calculation programs and calculation maps for the calculations in addition to various types of data, a random access memory (RAM) that temporarily stores calculation results, and the like.

As shown in FIG. **1**, the electronic control unit **300** is connected to the sensors described below.

An accelerator position sensor **301** detects the amount of an accelerator pedal depressed by a driver. An air flow meter **302** detects the amount and temperature of the air drawn into the internal combustion engine **400**. A crank angle sensor **303** detects the engine speed NE based on a rotation angle of a crankshaft, which is an output shaft of the internal combustion engine **400**. A turbine rotation number sensor **304** is arranged in the vicinity of the switching mechanism **120** and detects the rotation speed of a turbine of the torque converter **110**. A first pulley rotation speed sensor **305** is arranged in the vicinity of the first pulley **130** and detects the rotation speed Nin of the first pulley **130**. A second pulley rotation speed sensor **306** is arranged in the vicinity of the second pulley **150** and detects the rotation speed Nout of the second pulley **150**. Wheel speed sensors **307** are respectively arranged in the vicinity of the vehicle wheels and detect the rotation speed of the corresponding vehicle wheels. A temperature sensor **308** detects the temperature of the hydraulic oil supplied to the hydraulic pressure chambers **134** and **154** by the hydraulic pressure control unit **200**.

Based on output signals from the various sensors **301** to **308**, the electronic control unit **300** entirely controls the internal combustion engine **400** and the continuously variable transmission **100**. For example, a vehicle speed SPD is calculated based on the rotation speed Nout of the second pulley **150**, which is detected by the second pulley rotation speed sensor **306**. A required torque is calculated based on the current vehicle speed SPD and the depression amount of the accelerator pedal detected by the accelerator position sensor **301**. An opening degree Th of a throttle valve **411**, which is arranged in an intake passage **410** of the internal combustion engine **400**, is adjusted to adjust intake air amount GA and realize the required torque.

Further, when adjusting the intake air amount GA, the electronic control unit **300** calculates a target gear ratio γtrg as the gear ratio γ that most efficiently generates the request torque. The electronic control unit **300** also executes gear shift control that controls the hydraulic pressure control unit **200** so that the actual gear ratio γ conforms to the calculated target gear ratio γtrg. In other words, in the present embodiment, the electronic control unit **300** and the hydraulic pressure control unit **200** form the hydraulic pressure controller for the continuously variable transmission **100**.

In the gear shift control, the current gear ratio γ is calculated based on the rotation speed Nin of the first pulley **130** and the rotation speed Nout of the second pulley **150**. Further, to bring the gear ratio γ closer to the target gear ratio γtrg, the hydraulic pressure Pin in the first pulley **130** is changed to change the thrust Wpri. The thrust Wpri at the first pulley **130** is changed, and the hydraulic pressure Pout in the second pulley **150** is changed to change the thrust Wsec so that the belt **140** does not slip on the pulleys **130** and **150**.

A structure of the hydraulic pressure control unit **200** will now be described in detail with reference to FIG. **2**. FIG. **2** is a schematic diagram showing the structure of the hydraulic pressure control unit **200** in the hydraulic pressure controller according to the present embodiment.

As shown at the left side in FIG. **2**, the hydraulic pressure control unit **200** includes a regulator valve **212** that adjusts the pressure of the hydraulic oil discharged from the oil pump **211** to generate line pressure Pl, which becomes the originating pressure of the hydraulic pressures Pin and Pout. The regulator valve **212** sends some of the hydraulic oil discharged from the oil pump **211** to another regulator valve (not shown) based on the level of the line pressure Pl. The hydraulic oil send from the regulator valve **212** to another regulator valve is supplied to the torque converter **110** and the switching mechanism **120** as a hydraulic pressure Psec. The regulator valve **212** adjusts the line pressure Pl by discharging some of the hydraulic oil discharged from the oil pump **211** based on the level of the line pressure Pl.

The hydraulic pressure control unit **200** includes a modulator valve **214** that further depressurizes the line pressure Pl and generates a fixed modulator pressure Pm. The modulator pressure Pm output from the modulator valve **214** is supplied to a first solenoid valve **215** and a second solenoid valve **216**.

The first solenoid valve **215**, which is electrically driven by a drive command output from the electronic control unit **300**, adjusts the modulator pressure Pm to generate a first solenoid pressure Pslp, which is the driving hydraulic pressure of a first control valve **217**. More specifically, the first solenoid valve **215**, which is a normally open type solenoid valve that closes when supplied with power, outputs a larger first solenoid pressure Pslp as the drive duty decreases in accordance with the level of the drive duty output as the drive command from the electronic control unit **300**.

The second solenoid valve **216**, which is electrically driven by a drive command output from the electronic control unit **300**, adjusts the modulator pressure Pm to generate a second solenoid pressure Psls, which is the driving hydraulic pressure of a second control valve **218**. The second solenoid valve **216**, which is also a normally open type solenoid valve like the first solenoid valve **215**, outputs a larger second solenoid pressure Psls as the drive duty decreases in accordance with the level of the drive duty output as the drive command from the electronic control unit **300**.

The first solenoid pressure Pslp, which is output from the first solenoid valve **215**, is input to the first control valve **217**. The first control valve **217** adjusts the line pressure Pl in accordance with the first solenoid pressure Pslp. This adjusts the level of the hydraulic pressure supplied to the hydraulic pressure chamber **134** in the first pulley **130** and controls the hydraulic pressure Pin of the hydraulic pressure chamber **134**.

The first control valve **217** includes three input ports **217***d*, **217***e*, and **217***f*. The first solenoid pressure Pslp is input to the second input port **217***e*. The line pressure Pl is input to the first input port **217***d*, and the solenoid modulator pressure Psolmod is input to the third input port **217***f*.

The first control valve **217** accommodates a valve body **217***a* that is movable in the axial direction. A first pressure chamber **217***b* and a second pressure chamber **217***c* are defined and formed in the first control valve **217** so as to sandwich the valve body **217***a*. The second input port **217***e* is connected to the first pressure chamber **217***b*, and the third input port **217***f* is connected to the second pressure chamber **217***c*.

As a result, the first solenoid pressure Pslp supplied to the first pressure chamber **217***b* through the second input port **217***e* and the solenoid modulator pressure Psolmod supplied to the second pressure chamber **217***c* through the third input port **217***f* act from opposite directions on the valve body **217***a* of the first control valve **217**. A spring **217***g* is accommodated in the first pressure chamber **217***b* in a compressed state as a biasing member for biasing the valve body **217***a* toward the second pressure chamber **217***c*.

The solenoid modulator pressure Psolmod supplied to the second pressure chamber **217***c* through the third input port **217***f* is adjusted to a certain level by a pressure adjustment valve **230**, as will be described later. Thus, the balance of the forces acting on the valve body **217***a* of the first control valve **217** changes in accordance with the level of the first solenoid pressure Pslp supplied to the first pressure chamber **217***b* through the second input port **217***e*. The valve body **217***a* is moved in the axial direction in accordance with changes in the balance of the forces.

The first control valve **217** further includes an output port **217***h* connected to the hydraulic pressure chamber **134** in the first pulley **130** through a failsafe valve **219**, which will be described later, a discharge port **217***i* connected to a discharge passage, and a feedback port **217***j*.

The first control valve **217** moves the valve body **217***a* in accordance with the balance of the forces acting on the valve body **217***a*. Further, the first control valve **217** moves the valve body **217***a* toward the second pressure chamber **217***c* when the first solenoid pressure Pslp become high and the pressure in the first pressure chamber **217***b* becomes high. Thus, in the first control valve **217**, when the first solenoid pressure Pslp becomes high, the first input port **217***d* opens and communicates the first input port **217***d* and the output port **217***h*.

As a result, when the first solenoid pressure Pslp becomes high, some of the hydraulic oil input through the first input port **217***d* is supplied to the hydraulic pressure chamber **134** in the first pulley **130** through the output port **217***h*.

The first control valve **217** is formed so that the driving force toward the second pressure chamber **217***c*, that is, the driving force in the valve opening direction, increases as the input first solenoid pressure Pslp increases. Thus, as the first solenoid pressure Pslp increases, the hydraulic pressure output from the first control valve **217** increases and the hydraulic pressure Pin of the hydraulic pressure chamber **134** increases.

Further, as shown in FIG. **2**, some of the hydraulic pressure supplied to the hydraulic pressure chamber **134** is fed back to act on the valve body **217***a* through the feedback port **217***j*. Thus, when the hydraulic pressure Pin increases and approaches the hydraulic pressure corresponding to the level of the first solenoid pressure Pslp, the valve body **217***a* moves toward the first pressure chamber **217***b*. When the hydraulic pressure Pin becomes equal to the hydraulic pressure corresponding to the level of the first solenoid pressure Pslp, the valve body **217***a* closes the first input port **217***d*.

When the hydraulic pressure Pin becomes higher than the hydraulic pressure corresponding to the level of the first solenoid pressure Pslp, the valve body **217***a* further moves toward the first pressure chamber **217***b*, and the discharge port **217***i* opens and communicates the output port **217***h* and the discharge port **217***i*. This discharges the hydraulic oil from the hydraulic pressure chamber **134** to the discharge passage through the output port **217***h* and the discharge port **217***i*, and the hydraulic pressure Pin of the hydraulic pressure chamber **134** is adjusted to the hydraulic pressure corresponding to the level of the first solenoid pressure Pslp.

The hydraulic oil discharged through the discharge passage is collected in an oil pan (not shown) and supplied again to each part by the oil pump **211**.

The second solenoid pressure Psls output from the second solenoid valve **216** is input to the second control valve **218**. The second control valve **218** adjusts the line pressure Pl in accordance with the second solenoid pressure Psls to adjust the level of the hydraulic pressure supplied to the hydraulic pressure chamber **154** in the second pulley **150** and control the hydraulic pressure Pout of the hydraulic pressure chamber **154**.

The second control valve **218** includes three input ports **218***d*, **218***e*, and **218***f* like the first control valve **217**. The second solenoid pressure Psls is input to the second input port **218***e*. The line pressure Pl is input to the first input port **218***d*, and the solenoid modulator pressure Psolmod is input to the third input port **218***f*.

The second control valve **218** accommodates a valve body **218***a*, which is movable in the axial direction. A first pressure chamber **218***b* and a second pressure chamber **218***c* are defined and formed in the second control valve **218** so as to sandwich the valve body **218***a*. The second input port **218***e* is connected to the first pressure chamber **218***b*, and the third input port **218***f* is connected to the second pressure chamber **218***c*.

As a result, the second solenoid pressure Psls supplied to the first pressure chamber **218***b* through the second input port **218***e* and the solenoid modulator pressure Psolmod supplied to the second pressure chamber **218***c* through the third input port **218***f* act from opposite directions on the valve body **218***a* of the second control valve **218**. A spring **218***g* is accommodated in the first pressure chamber **218***b* in a compressed state as a biasing member for biasing the valve body **218***a* toward the second pressure chamber **218***c*.

The solenoid modulator pressure Psolmod supplied to the second pressure chamber **218***c* through the third input port **218***f* is adjusted to a certain level by the pressure adjustment valve **230**, as will be described later. Thus, the balance of the forces acting on the valve body **218***a* of the second control valve **218** changes in accordance with the level of the second solenoid pressure Psls supplied to the first pressure chamber **218***b* through the second input port **218***e*. The valve body **218***a* is moved in the axial direction in accordance with such change in balance of the forces.

The second control valve **218** further includes an output port **218***h* connected to the hydraulic pressure chamber **154** in the second pulley **150**, a discharge port **218***i* connected to the discharge passage, and a feedback port **218***j*.

The second control valve **218** moves the valve body **217***a* in accordance with the balance of forces acting on the valve body **218***a*. When the second solenoid pressure Psls increases and the pressure in the first pressure chamber **218***b* increases, the valve body **218***a* is moved toward the second pressure chamber **218***c*. Thus, in the second control valve **218**, when the second solenoid pressure Psls becomes high, the first input port **218***d* opens and communicates the first input port **218***d* and the output port **218***h*.

Thus, when the second solenoid pressure Psls becomes high, some of the hydraulic oil input through the first input port **218***d* is supplied to the hydraulic pressure chamber **154** in the second pulley **150** through the output port **218***h*.

The second control valve **218** is formed so that the driving force toward the second pressure chamber **218***c*, that is, the driving force in the valve opening direction increases as the input second solenoid pressure Psls increases. Thus, as the second solenoid pressure Psls increases, the hydraulic pressure output from the second control valve **218** increases and the hydraulic pressure Pout of the hydraulic pressure chamber **154** increases.

Further, as shown in FIG. **2**, some of the hydraulic pressure supplied to the hydraulic pressure chamber **154** is fed back to act on the valve body **218***a* through the feedback port **218***j*. Thus, when the hydraulic pressure Pout increases and approaches the hydraulic pressure corresponding to the level of the second solenoid pressure Psls, the valve body **218***a* moves toward the first pressure chamber **218***b*. When the hydraulic pressure Pout becomes equal to the hydraulic pressure corresponding to the level of the second solenoid pressure Psls, the valve body **218***a* closes the first input port **218***d*.

When the hydraulic pressure Pout becomes higher than the hydraulic pressure corresponding to the level of the second solenoid pressure Psls, the valve body **218***a* is further moved toward the first pressure chamber **218***b*, and the discharge port **218***i* opens thereby communicating the output port **218***h* and the discharge port **218***i*. This discharges the hydraulic oil in the hydraulic pressure chamber **154** to the discharge passage through the output port **218***h* and the discharge port **218***i*, and the hydraulic pressure Pout of the hydraulic pressure chamber **154** is adjusted to the hydraulic pressure corresponding to the level of the second solenoid pressure Psls.

The solenoid modulator pressure Psolmod is adjusted to a certain level through the pressure adjustment valve **230** shown at the upper part of FIG. **2**.

The pressure adjustment valve **230** includes an input port **230***b*, an output port **230***c*, and a discharge port **230***d*. The pressure adjustment valve **230** accommodates a valve body **230***a*, which is movable in an axial direction.

A spring **230***e* is accommodated in the pressure adjustment valve **230** in a compressed state as a biasing member for biasing the valve body **230***a* in one direction. Thus, the valve body **230***a* is always biased in the same direction by the spring **230***e* so as to communicate the input port **230***b* and the output port **230***c* as shown in FIG. **2**.

Hydraulic pressure further depressurized from the line pressure Pl is input to the input port **230***b* of the pressure adjustment valve **230**. The output port **230***c* of the pressure adjustment valve **230** is connected to the third input port **217***f* of the first control valve **217** and the third input port **218***f* of the second control valve **218** as shown in FIG. **2**.

Some of the hydraulic oil output from the output port **230***c* is fed back to the valve body **230***a* through a feedback port **230***f*.

Thus, when the solenoid modulator pressure Psolmod, which is the hydraulic pressure of the hydraulic oil output from the output port **230***c*, becomes excessively high, the valve body **230***a* is moved against the biasing force of the spring **230***e* by the hydraulic pressure fed back through the feedback port **230***f*.

In this manner, when the valve body **230***a* is moved in the valve opening direction against the biasing force of the spring **230***e*, the valve body **230***a* closes the input port **230***b* and opens the discharge port **230***d* thereby communicating the output port **230***c* and the discharge port **230***d*. As a result, some of the hydraulic oil supplied to the second pressure chamber **217***c* of the first control valve **217** and the second pressure chamber **218***c* of the second control valve **218** is discharged through the output port **230***c* and the discharge port **230***d*.

When some of the hydraulic oil supplied to the second pressure chambers **217***c* and **218***c* through the discharge port **230***d* of the pressure adjustment valve **230** is discharged in such a manner, the solenoid module pressure Psolmod decreases and adjusts the solenoid modulator pressure Psolmod to a certain level.

The hydraulic oil discharged from the discharge port **230***d* is collected in the oil pan (not illustrated) through the discharge passage and supplied to each part again by the oil pump **211**.

The electronic control unit **300** executes gear shift control to change the drive duty output to the first solenoid valve **215** and the second solenoid valve **216** and control the first solenoid pressure Pslp and the second solenoid pressure Psls.

The electronic control unit **300** controls the hydraulic pressure control unit **200** to adjust the hydraulic pressures Pin and Pout of the hydraulic pressure chambers **134** and **154** in the pulleys **130** and **150** so that the gear ratio γ conforms to the target gear ratio γtrg.

When an abnormality occurs in the first control valve **217** or the first solenoid valve **215**, the hydraulic pressure Pin cannot be properly controlled and the hydraulic pressure Pin may increase one-sidedly or decrease one-sidedly.

For instance, when a foreign matter or the like is caught in the first control valve **217** and the necessary amount of hydraulic oil cannot be supplied to the hydraulic pressure chamber **134**, the hydraulic pressure Pin becomes insufficient and imbalances the thrusts Wpri and Wsec in the pulleys **130** and **150**, and the tension of the belt **140** pushes and opens the first pulley **130**. As a result, the gear ratio γ becomes high in a one-sided manner and the engine speed NE increases.

In this regard, the hydraulic pressure control unit **200** includes the failsafe valve **219**, which switches the supply path of the hydraulic oil supplied to the first pulley **130**.

The hydraulic oil output from the output port **217***h* of the first control valve **217** is supplied to the hydraulic pressure chamber **134** in the first pulley **130** through the failsafe valve **219** as shown at the lower right side of FIG. **2**. In the first pulley **130**, the movable sheave moves in accordance with the hydraulic pressure Pin in the hydraulic pressure chamber **134** as described above. This changes the distance between the fixed sheave and the movable sheave.

The hydraulic oil output from the output port **218***h* of the second control valve **218** and supplied to the hydraulic pressure chamber **154** in the second pulley **150** is directly supplied to the hydraulic pressure chamber **154** in the second pulley **150** without passing the failsafe valve **219**. In the second pulley **150**, the movable sheave is moved in accordance with the hydraulic pressure Pout in the hydraulic pressure chamber **154** as described above. This changes the distance between the fixed sheave and the movable sheave.

The failsafe valve **219** arranged between the first control valve **217** and the first pulley **130** includes a first input port **219***a* to which the hydraulic oil output from the output port **217***h* of the first control valve **217** is introduced and a second input port **219***b* into which the hydraulic oil output from the output port **218***h* of the second control valve **218** is drawn as shown in FIG. **2**. The failsafe valve **219** is formed to selectively communicate one of the first input port **219***a* and the second input port **219***b* to the output port **219***c* in accordance with the position of the valve body driven by the driving hydraulic pressure output from a switching solenoid valve **220**.

More specifically, in a normal state in which the switching solenoid valve **220** is in the "OFF" state and the driving hydraulic pressure is not output from the switching solenoid valve **220**, the first input port **219***a* is in communication with the output port **219***c*. In a fail state in which the switching solenoid valve **220** is in the "ON" state and the driving hydraulic pressure is output from the switching solenoid valve **220**, the second input port **219***b* is in communication with the output port **219***c*. In other words, the failsafe valve **219** is formed to select one of the hydraulic oil adjusted through the first control valve **217** and the hydraulic oil adjusted through the second control valve **218** and output the selected one to the first pulley **130**.

The electronic control unit **300** switches the switching solenoid valve **220** to the "ON" state when the hydraulic pressure Pin in the first pulley **130** cannot be properly controlled by the first control valve **217**. This switches the supply path of the hydraulic oil so that the hydraulic oil adjusted by the second control valve **218** is also supplied to the first pulley **130**. The hydraulic pressures Pin and Pout of the pulleys **130** and **150** thus become equal, and the gear ratio γ can be suppressed from increasing in a one-sided manner.

As shown in the right side of FIG. **2**, the oil path for drawing hydraulic oil into the hydraulic pressure chamber **134** of the first pulley **130** and the oil path for drawing the hydraulic oil into the hydraulic pressure chamber **154** of the second pulley **150** each include an orifice. These orifices are used so that the hydraulic oil in the hydraulic pressure chambers **134** and **154** is not rapidly discharged and the hydraulic pressures Pin and Pout are not rapidly decreased to prevent slipping of the belt **140** on the pulleys **130** and **150**.

In the hydraulic pressure controller of the present embodiment, the line pressure Pl is maintained at the minimum required level, and the feedback adjustment of the line pressure Pl is performed to minimize the drive load on the oil pump **211**.

More specifically, the hydraulic pressure Pin of the hydraulic pressure chamber **134** in the first pulley **130** and the second solenoid pressure Psls output by the second solenoid valve **216** are each conveyed to a reduction valve **213** to feedback adjust the line pressure Pl. The reduction valve **213** adjusts the modulator pressure Pm in accordance with the conveyed hydraulic pressure Pin and the second solenoid pressure Psls to generate a line pressure adjustment hydraulic pressure Psrv. The line pressure adjustment hydraulic pressure Psrv is conveyed to the regulator valve **212** and used to adjust the line pressure Pl in the regulator valve **212**.

In other words, in the hydraulic pressure control unit **200** of the present embodiment, the line pressure Pl is feedback adjusted in accordance with the line pressure adjustment hydraulic pressure Psrv that changes in accordance with the levels of the hydraulic pressures Pin and Pout in the pulleys **130** and **150**.

Through such feedback adjustment with the reduction valve **213**, the line pressure Pl is adjusted to be slightly higher than the higher one of the hydraulic pressure Pin and the hydraulic pressure Pout.

The feedback adjustment of the line pressure Pl in accordance with the levels of the hydraulic pressures Pin and Pout supplied to the pulleys **130** and **150** prevents the line pressure Pl from increasing more than necessary and prevents the drive load of the oil pump **211** from becoming excessively high.

When changing the hydraulic pressures Pin and Pout, the valve bodies **217***a* and **218***a* of the control valves **217** and **218** are driven and the valve body of the regulator valve **212** is driven. However, inertia may excessively move the valve body immediately after starting to drive the valve body. This may oscillate the valve body.

For instance, immediately after decreasing the drive duty of the first solenoid valve **215** and increasing the first solenoid pressure Pslp, which is the driving hydraulic pressure, to increase the hydraulic pressure Pin as shown in FIG. **3**, inertia excessively moves the valve body **217***a* in the valve opening direction and oscillates the valve body **217***a*. When the valve body **217***a* is oscillated, the oscillation of the valve body **217***a* may also oscillate the hydraulic pressure Pin adjusted through the control valve **217** over the target hydraulic pressure Ptrg as shown in FIG. **3**.

When the hydraulic pressure Pin is oscillated in such a manner, the tension on the belt **140** repetitively increases and decreases therewith. This may result in the belt **140** slipping on the pulleys **130** and **150** or excessive load being applied to the belt **140** thereby lowering the durability of the continuously variable transmission **100**.

Once the hydraulic pressure Pin is oscillated as shown in FIG. **3** when the hydraulic pressure Pin is greater than the hydraulic pressure Pout, the line pressure adjustment hydraulic pressure Psrv that changes in accordance with the oscillating hydraulic pressure Pin is input to the regulator valve **212**. Thus, the line pressure Pl is feedback adjusted in accordance with changes in the oscillating hydraulic pressure Pin, and the line pressure Pl adjusted by the regulator valve **212** is oscillated. In other words, an adverse cycle in which the oscillation is propagated to the line pressure Pl, and the hydraulic pressures Pin and Pout supplied to the pulleys **130** and **150** are adjusted in accordance with the oscillating line pressure Pl occurs.

As a result, the oscillation of the hydraulic pressure Pin resists attenuation, and a state in which the hydraulic pressures Pin and Pout supplied to the pulleys **130** and **150** oscillate is continued for a long period of time.

In the electronic control unit **300** of the present embodiment, oscillation suppression control is executed to suppressing the oscillation of the hydraulic pressure Pin by decreasing the drive duty and increasing the first solenoid pressure Pslp and then further decreasing the drive duty and increasing the first solenoid pressure Pslp as shown in FIG. **4**.

When the drive duty is decreased at time t**1** as shown in FIG. **3** to increase the hydraulic pressure Pin, the hydraulic pressure Pin exceeds and overshoots the target hydraulic pressure Ptrg. As shown in FIG. **3**, the hydraulic pressure Pin starts to decrease from time t**3** and starts to increase again at time t**4**. In the oscillation suppression control, as shown in FIG. **4**, the drive duty is decreased at time t**1**. Then, at time t**2**, the drive duty is further decreased and the first solenoid pressure Pslp is further increased so that the valve body **217***a* can be biased in the valve opening direction from time t**3** to time t**4**.

In this manner, by further increasing the first solenoid pressure Pslp at time t**2**, the valve body **217***a* excessively moved in the valve opening direction as the first solenoid pressure Pslp increases at time t**1** offsets the portion of the force acting to move the valve body **217***a* in the valve closing direction and suppresses oscillation of the valve body **217***a*.

This suppresses undershooting of the hydraulic pressure Pin at time t**4** and suppresses oscillation of the hydraulic pressure Pin as show in FIG. **4**.

The flow of processes related to the oscillation suppression control will now be specifically described with reference to FIGS. **5** and **6**. FIG. **5** is a flowchart showing the flow of a series of processes related to the oscillation suppression control, and FIG. **6** is a flowchart showing the flow of a process of the oscillation suppression control. The series of processes shown in FIG. **5** are repeatedly executed in predetermined control cycles in the electronic control unit **300** when the engine is running.

When the series of processes shown in FIG. **5** is started, the electronic control unit **300** first determines in step S**10** whether or not to increase the hydraulic pressure Pin. For instance, if the target gear ratio γtrg is less than the current gear ratio γ and the hydraulic pressure Pin needs to be increased for upshifting, it is determined that the hydraulic pressure Pin should be increased. If the target gear ratio γtrg and the current gear ratio γ are in conformance or if the target gear ratio γtrg is greater than the current gear ratio γ, it is determined that the hydraulic pressure Pin should not be increased.

When determined in step S**10** that the hydraulic pressure Pin should be increased (step S**10**: YES), the electronic control unit **300** proceeds to step S**20** and determines whether or not the hydraulic pressure Pin is greater than the hydraulic pressure Pout.

When determined in step S**20** that the hydraulic pressure Pin is greater than the hydraulic pressure Pout (step S**20**: YES), the electronic control unit **300** proceeds to step S**100** and executes the oscillation suppression control shown in FIG. **6**.

When the oscillation suppression control is started, as shown in FIG. **6**, in step S**110**, the electronic control unit **300** first sets the timing for further increasing the first solenoid pressure Pslp based on the drive duty of the first solenoid valve **215** and the temperature of hydraulic oil detected by the oil temperature sensor **308**.

After the output of the first solenoid pressure Pslp for increasing the hydraulic pressure Pin is started and the valve body **217***a* is driven in the valve opening direction, the timing for starting movement of the valve body **217***a* in the valve closing direction, that is, the timing of time t**3** in FIG. **3** changes in accordance with the responsiveness or the like of the actual movement of the valve body **217***a* with respect to changes in the first solenoid pressure Pslp.

When the temperature of the hydraulic oil is high, the viscosity of the hydraulic oil decreases. Thus, the responsiveness of the actual movement of the valve body **217***a* with respect to changes in the first solenoid pressure Pslp increases as the temperature of the hydraulic oil increases. That is, as the temperature of the hydraulic oil increases, the valve body **217***a* moves more rapidly when the first solenoid pressure Pslp is changed. Further, after the first solenoid pressure Pslp is output, the timing at which the valve body **217***a* starts to move in the valve closing direction is also advanced as the temperature of the hydraulic oil increases.

As the first solenoid pressure Pslp increase to increase the hydraulic pressure Pin, the valve opening speed of the valve body **217***a* increases when the first solenoid pressure Pslp is output. That is, as the first solenoid pressure Pslp increases to increase the hydraulic pressure Pin, the valve body **217***a* moves more rapidly, and the timing at which the valve body **217***a* starts to move in the valve closing direction is also advanced.

Here, the time (time Tint in FIG. **4**) from when output of the first solenoid pressure Pslp starts to when the first solenoid pressure Pslp is further increased is set based on the temperature of the hydraulic oil and the level of the drive duty when output of the first solenoid pressure Pslp is started to increase the hydraulic pressure Pin. The time Tint is set to be shorter as the temperature of the hydraulic oil increases and the drive duty decreases when output of the first solenoid pressure Pslp for increasing the hydraulic pressure Pin is started.

By setting the length of the time Tint in this manner, the timing at which the first solenoid pressure Pslp is further increased is advanced as the temperature of the hydraulic oil increases and the first solenoid pressure Pslp increases to increase the hydraulic pressure Pin.

After setting the timing for further increasing the first solenoid pressure Pslp in step S**110**, the electronic control unit **300** proceeds to step S**120** and further increases the first solenoid pressure Pslp when the set timing is reached. Specifically, the drive duty is further decreased when the set timing is reached. This further increases the first solenoid pressure Pslp.

When the oscillation suppression control is executed through step S**110** and step S**120** in this manner, the electronic control unit **300** temporarily terminates the series of processes.

If determined in step S**10** of FIG. **5** that the hydraulic pressure Pin should not be increased (step S**10**: NO), the electronic control unit **300** skips step S**20** and step S**100** and terminates the series of processes without executing the oscillation suppression control.

If determined in step S**20** that the hydraulic pressure Pin is less than or equal to the hydraulic pressure Pout (step S**20**: NO), the electronic control unit **300** skips step S**100** and terminates the series of processes without executing the oscillation suppression control.

In this manner, after increasing the first solenoid pressure Pslp to increase the hydraulic pressure Pin, the oscillation suppression control further temporarily increases the first solenoid pressure Pslp to offset the portion of the force that acts to move the valve body **217***a*, which has been excessively moved in the valve opening direction, in the valve closing direction. This suppresses movement of the valve body **217***a*. Thus, as shown in FIG. **4**, undershooting of the hydraulic pressure Pin can be suppressed, and oscillation of the hydraulic pressure Pin can be suppressed.

The first embodiment has the advantages described below.

(1) In the hydraulic pressure controller according to the present embodiment, after driving the valve body **217***a* of the first control valve **217** that controls the hydraulic pressure Pin in the valve opening direction, the oscillation suppression control is executed to temporarily suppress the movement of the valve body **217***a*.

Thus, oscillation of the valve body **217***a* when the valve body **217***a* is driven in the valve opening direction is suppressed, and repetitive increasing and decreasing of the hydraulic pressure Pin supplied to the first pulley **130** over the target hydraulic pressure Ptrg is suppressed. Further, oscillation of the valve body **217***a* is suppressed in this manner, and oscillation of the hydraulic pressure Pin supplied to the first pulley **130** is suppressed. This suppresses oscillation of the line pressure Pl, which is feedback adjusted based on the hydraulic pressure Pin.

As a result, the occurrence of an adverse cycle is suppressed in which the line pressure Pl is adjusted based on the oscillating hydraulic pressure Pin, and the hydraulic pressures Pin and Pout supplied to the pulleys **130** and **150** are adjusted based on the oscillating line pressure Pl.

In other words, the line pressure Pl is feedback adjusted in accordance with the levels of the hydraulic pressures Pin and Pout, excessive increase of the drive load of the oil pump **211** is suppressed, and oscillation of the hydraulic pressures Pin and Pout and the line pressure Pl when the hydraulic pressure Pin changes is suppressed.

Further, as a result, when the hydraulic pressure Pin of the first pulley **130** changes, repetitive increase and decrease of the tension on the belt **140** is suppressed, and a decrease in the durability of the continuously variable transmission **100** is suppressed.

(2) As described above, after the first solenoid pressure Pslp is output to increase the hydraulic pressure Pin and the valve body **217***a* is driven in the valve opening direction, the timing at which the valve body **217***a* starts to move in the valve closing direction changes in accordance with the responsiveness or the like of the actual movement of the valve body **217***a* with respect to changes in the first solenoid pressure Pslp. When the temperature of the hydraulic oil is high, the viscosity of the hydraulic oil decreases. Thus, the responsiveness of the actual movement of the valve body **217***a* with respect to changes in the first solenoid pressure Pslp increases as the temperature of the hydraulic oil increases. In other words, as the temperature of the hydraulic oil increases, the valve body **217***a* moves more readily when the first solenoid pressure Pslp. Further, as the temperature of the hydraulic oil increases, the timing at which the valve body **217***a* starts to move in the valve closing operation becomes earlier.

In the first embodiment, after starting the output of the first solenoid pressure Pslp to drive the valve body **217***a* in the valve opening direction thereby increasing the hydraulic pressure Pin, the timing for further increasing the first solenoid pressure Pslp is advanced as the temperature of the hydraulic oil increases. Thus, the first solenoid pressure Pslp can be changed in accordance with changes in the responsiveness of the actual movement of the valve body **217***a* with respect to the changes in the first solenoid pressure Pslp when the temperature of the hydraulic oil changes. Accordingly, the oscillation of the valve body **217***a* is accurately suppressed, and the oscillation of the hydraulic pressure Pin is properly suppressed.

(3) The valve opening speed of the valve body **217***a* when the first solenoid pressure Pslp increases as the first solenoid pressure Pslp increases to increase the hydraulic pressure Pin. In other words, as the first solenoid pressure Pslp output to drive the valve body **217***a* in the valve opening direction increases, the valve body **217***a* is moved more readily when the first solenoid pressure Pslp is output. This advances the timing at which the valve body **217***a* starts to move in the valve closing direction. In this regard, in the first embodiment, the timing for further increasing the first solenoid pressure Pslp is set based on the level of the drive duty output to increase the hydraulic pressure Pin. As the duty ratio output to increase the hydraulic pressure Pin decreases and the first solenoid pressure Pslp output to increase the hydraulic pressure Pin increases, the timing for changing the drive duty to further increase the first solenoid pressure Pslp is advanced.

Thus, the timing for changing the first solenoid pressure Pslp can be set in accordance with changes in the valve opening speed of the valve body **217***a* when the first solenoid pressure Pslp is output, and oscillation of the valve body **217***a* can be effectively suppressed.

(4) The oscillation of the valve body **217***a* is apt to occurring when the first solenoid pressure Pslp is output to increase the hydraulic pressure Pin supplied to the first pulley **130**, and the first solenoid pressure Pslp acts on the valve body **217***a* to drive the valve body **217***a* in the valve opening direction. In this regard, in the first embodiment, the oscillation suppression control is executed when the hydraulic pressure Pin increases and when the hydraulic pressure Pin is greater than the hydraulic pressure Pout.

Thus, when increasing the hydraulic pressure Pin supplied to the first pulley **130**, the oscillation suppression control is executed under a situation that may result in an adverse cycle in which the hydraulic pressures Pin and Pout supplied to the pulley **130** and **150** are oscillated continues for a long period of time due to feedback adjustment of the line pressure P**1** performed in accordance with changes in the hydraulic pressure Pin supplied to the first pulley **130**.

Accordingly, a state in which the hydraulic pressures Pin and Pout supplied to the pulleys **130** and **150** oscillate continuously over a long period of time is suppressed in a preferred manner.

The first embodiment may be modified in the forms described below.

The first embodiment is configured to set the timing that further increases the first solenoid pressure Pslp based on the temperature of the hydraulic oil and the drive duty for increasing the hydraulic pressure Pin. The timing for further increasing the first solenoid pressure Pslp may also be set based on only one of the temperature of the hydraulic oil and the drive duty.

The method for setting the timing to further increase the first solenoid pressure Pslp in the first embodiment is one example of a method for setting the timing for further increasing the first solenoid pressure Pslp, and the present invention is not limited to the setting shown illustrated in the first embodiment. In other words, as long as movement of the valve body **217***a* in the valve closing direction can be suppressed after the first solenoid pressure Pslp is output to increase the hydraulic pressure Pin, the method for setting the timing for further increasing the first solenoid pressure Pslp can be changed as required.

In the first embodiment, the oscillation suppression control increases the first solenoid pressure Pslp only once after the output of the first solenoid pressure Pslp is started to increase the hydraulic pressure Pin. However, the oscillation suppression control may increase the first solenoid pressure Pslp over a number of times. In other words, the oscillation suppression control may cyclically increase in accordance with the cycle of the oscillation of the hydraulic pressure Pin to suppress movement of the valve body **217***a* and suppress oscillation of the hydraulic pressure Pin.

In the first embodiment, the drive duty is changed to further increase the first solenoid pressure Pslp and thereby suppress undershooting in which the hydraulic pressure Pin becomes lower than the target hydraulic pressure Ptrg. In this regard, after increasing the first solenoid pressure Pslp to increase the hydraulic pressure Pin, the drive duty may be temporarily changed to decrease the first solenoid pressure Pslp and suppress overshooting in which the hydraulic pressure Pin becomes greater than the target hydraulic pressure Ptrg by suppressing movement of the valve body **217***a*. This suppresses oscillation of the hydraulic pressure Pin.

In addition, after increasing the first solenoid pressure Pslp to increase the hydraulic pressure Pin, the first solenoid pressure Pslp may be alternately and repetitively increased and decreased in accordance with the oscillation cycle of the hydraulic pressure Pin to temporarily suppress movement of the valve body **217***a* and suppress oscillation of the hydraulic pressure Pin.

Second Embodiment

A second embodiment of a hydraulic pressure controller for a continuously variable transmission according to the present invention applied to an electronic control unit **300** that controls a continuously variable transmission **100**, which is installed in a vehicle, and a hydraulic pressure control unit **200** will now be described with reference to FIGS. **7** and **8**. The present embodiment differs from the first embodiment in that a closing valve **240** and a switching solenoid valve **241**, which drives the closing valve **240**, are added to the hydraulic pressure control unit **200**, as shown in FIG. **7**. Otherwise, the present embodiment is the same as the first embodiment. Thus, in the description hereafter, components that are the same as the first embodiment will not be described, and components differing from the first embodiment will be described in detail.

The hydraulic pressure control unit **200** of the present embodiment includes the closing valve **240** on the discharge passage through which the hydraulic oil discharged from the discharge port **230***d* of the pressure adjustment valve **230** flows, as shown in FIG. **7**. The closing valve **240** can switch between states closing and opening the discharge passage.

Further, the hydraulic pressure control unit **200** of the present embodiment includes the switching solenoid valve **241** that drives the closing valve **240**. The switching solenoid valve **241** is a solenoid valve electrically driven based on a drive command from the electronic control unit **300** and switched between an "ON" state, in which the driving hydraulic pressure is output to the closing valve **240**, and an "OFF" state, in which the driving hydraulic pressure is not output.

When the switching solenoid valve **241** is switched to the "ON" state, the driving hydraulic pressure is output, and the driving hydraulic pressure switches the closing valve **240** to the state closing the discharge passage, that is, a valve closing state.

When the switching solenoid valve **241** is switched to the "OFF" state, the drive hydraulic pressure is not output, and the closing valve **240** is switched to the state opening the discharge passage, that is, the valve opening state.

In the present embodiment, the discharge passage is closed by closing the closing valve **240** to prohibit the discharge of hydraulic oil from the second pressure chamber **217***c* of the first control valve **217** and the supply of hydraulic oil to the second pressure chamber **217***c* and execute oscillation suppression control that suppresses movement of the valve body **217***a*.

More specifically, the series of processes shown in FIG. **5** are repetitively executed while the engine is running in the same manner as the first embodiment. When determined that the hydraulic pressure Pin should be increased (step S**10**: YES) and that the hydraulic pressure Pin is greater than the hydraulic pressure Pout (step S**20**: YES), the process proceeds to step S**100** to execute the oscillation suppression control.

In the present embodiment, the oscillation suppression control shown in FIG. **8** is executed. As shown in FIG. **8**, when the oscillation suppression control is started, in step S**130**, the electronic control unit **300** sets the switching solenoid valve **241** to the "ON" state and switches the closing valve **240** to the valve closing state.

When closing the closing valve **240** in step S**130** and executing the oscillation suppression control, the electronic control unit **300** temporarily terminates the series of processes.

In this manner, when the closing valve **240** is closed in step S**130** and the oscillation suppression control is executed, the discharge of hydraulic oil from the discharge port **230***d* of the pressure adjustment valve **230** is prohibited. When the discharge of the hydraulic oil from the discharge port **230***d* of the pressure adjustment valve **230** is prohibited, the hydraulic oil in the second pressure chamber **217***c* is not discharged even if the solenoid modulator pressure Psolmod supplied to the second pressure chamber **217***c* of the first control valve **217** becomes high. When the hydraulic oil in the second pressure chamber **217***c* is no longer discharged, the solenoid modulator pressure Psolmod increases and the input port **230***b* is closed by the valve body **230***a*. This prohibits the supply of hydraulic oil to the second pressure chamber **217***c*.

Thus, the execution of the oscillation suppression control prohibits the discharge of hydraulic oil from the second pressure chamber **217***c* and the supply of hydraulic oil to the second pressure chamber **217***c*. As a result, the volume of the second pressure chamber **217***c* barely changes, and the valve body **217***a* barely moves.

Accordingly, when the output of the first solenoid pressure Pslp for increasing the hydraulic pressure Pin is started and the valve body **217***a* is driven in the valve opening direction, excessive movement of the valve body **217***a* in the valve opening direction can be suppressed. Further, subsequent movement of the valve body **217***a* can also be suppressed.

Thus, oscillation of the valve body **217***a* can be suppressed, and oscillation of the hydraulic pressure Pin can be suppressed when the valve body **217***a* is oscillated.

In this manner, the second embodiment has advantages (1) and (4) of the first embodiment.

The second embodiment may be modified in the forms described below.

In the second embodiment, the closing valve **240** is arranged in the discharge passage through which the hydraulic oil discharged from the discharge port **230***d* of the pressure adjustment valve **230** flows. The closing valve **240** only needs to be arranged at a position at which the discharge of hydraulic oil from the second pressure chamber **217***c* of the first control valve **217** and the supply of the hydraulic oil to the second pressure chamber **217***c* can be prohibited. Thus, the closing valve **240** may be arranged in a passage connecting the third input port **217***f* of the first control valve **217** and the output port **230***c* of the pressure adjustment valve **230**.

The closing valve **240** is driven by the driving hydraulic pressure output from the switching solenoid valve **241**. However, the closing valve **240** may be formed by an electrically driven solenoid valve, and the closing valve **240** may be directly driven by the electronic control unit **300**.

Elements that can be changed in each embodiment described include the following.

In the embodiments described above, the second solenoid pressure Psls is conveyed to the reduction valve **213**. In the reduction valve **213**, the line pressure adjustment hydraulic pressure Psrv is set based on the hydraulic pressure Pin and the second solenoid pressure Psls, and output to the regulator valve **212**. However, the hydraulic pressure Pout supplied to the hydraulic pressure chamber **154** in the second pulley **150** may be conveyed to the reduction valve **213** instead of the second solenoid pressure Psls, and the line pressure Pl may be feedback adjusted based on the hydraulic pressure Pin and the hydraulic pressure Pout.

The present invention may be applied even to a hydraulic pressure controller that feedback adjusts the line pressure Pl based on the hydraulic pressure Pin and the hydraulic pressure Pout.

When the line pressure Pl is feedback adjusted based on the hydraulic pressure Pin and the hydraulic pressure Pout, oscillation of the hydraulic pressure Pout as a result of the oscillation of the valve body **218***a* produces an advertent cycle in which the oscillation is propagated to the line pressure Pl and the hydraulic pressures Pin and Pout are adjusted based on the oscillating line pressure Pl.

Thus, when employing such a configuration, it is preferred that the oscillation suppression control be executed to further increase the second solenoid pressure Psls after increasing the second solenoid pressure Psls to increase the hydraulic pressure Pout.

When employing a configuration executing the oscillation suppression control that closes the closing valve **240** like in the second embodiment, the discharge of the hydraulic oil from the second pressure chamber **218***c* of the second control valve **218** and the supply of the hydraulic oil to the second pressure chamber **218***c* are prohibited during execution of the oscillation suppression control. Thus, oscillation of the valve body **218***a* of the second control valve **218** can be suppressed by executing oscillation suppression control that closes the closing valve **240** like in the second embodiment.

Thus, if the line pressure Pl is feedback adjusted based on the hydraulic pressure Pin and the hydraulic pressure Pout, it is preferred that the conditions for executing of the oscillation suppression control in the second embodiment be changed so that the oscillation suppression control is executed even when the hydraulic pressure Pout is increased.

The executing conditions of the oscillation suppression control are not limited to the executing conditions of the embodiments described above. Thus, the executing conditions of the oscillation suppression control can be changed as required in accordance with the configuration of the hydraulic pressure control unit **200** to which the present invention is applied.

In the embodiment described above, the hydraulic pressure controller for the continuously variable transmission according to the present invention is embodied as the hydraulic pressure controller for controlling the continuously variable transmission **100** that is installed in a vehicle. However, the present invention is not limited to a hydraulic pressure controller that controls a continuously variable transmission installed in a vehicle. That is, the present invention can be applied as a hydraulic pressure controller that controls a continuously variable transmission other than one installed in a vehicle.

In the embodiments described above, the oil temperature sensor **308** is used as an estimating means for estimating the temperature of the hydraulic oil, and the temperature of the hydraulic oil is detected by the oil temperature sensor **308**. However, the structure of the estimating means may be changed as long as the temperature of the hydraulic oil can be estimated. For instance, a structure that estimates a heat generation amount of the internal combustion engine **400** based on an integrated value of the intake air amount GA and estimates the temperature of the hydraulic oil based on the heat generation amount or a structure that estimates the temperature of the hydraulic oil based on the temperature of an engine coolant, which cools the internal combustion engine **400**, may be employed as the estimating means.

The structures of the continuously variable transmission **100**, the hydraulic pressure control unit **200**, and the electronic control unit **300** in the above embodiments are examples embodying the present invention. These structures may be changed as required.

In other words, the present invention is not limited to the continuously variable transmission **100**, the hydraulic pressure control unit **200**, and the electronic control unit **300** configured like in the embodiments described above. The present invention may be applied to a hydraulic pressure controller that feedback adjusts the line pressure Pl based on the hydraulic pressures Pin and Pout supplied to the pulleys **130** and **150**

DESCRIPTION OF THE REFERENCE CHARACTERS

100: continuously variable transmission
110: torque converter
120: switching mechanism
121: forward clutch
122: reverse brake
130: first pulley
134: hydraulic pressure chamber
140: belt
150: second pulley
154: hydraulic pressure chamber
200: hydraulic pressure control unit
211: oil pump
212: regulator valve
213: reduction valve
214: modulator valve
215: first solenoid valve
216: second solenoid valve
217: first control valve
217*a*: valve body
217*b*: first pressure chamber
217*c*: second pressure chamber
217*d*: first input port
217*e*: second input port
217*f*: third input port
217*g*: spring
217*h*: output port
217*i*: discharge port
217*j*: feedback port
218: second control valve
218*a*: valve body
218*b*: first pressure chamber
218*c*: second pressure chamber
218*d*: first input port
218*e*: second input port
218*f*: third input port
218*g*: spring
218*h*: output port
218*i*: discharge port
218*j*: feedback port
219: failsafe valve
219*a*: first input port
219*b*: second input port
219*c*: output port
220: switching solenoid valve
230: pressure adjustment valve
230*a*: valve body
230*b*: input port
230*c*: output port
230*d*: discharge port
230*e*: spring
230*f*: feedback port
240: closing valve
241: switching solenoid valve
300: electronic control unit
301: acceleration position sensor
302: airflow meter
303: crank angle sensor
304: turbine rotation speed sensor
305: first pulley rotation speed sensor
306: second pulley rotation speed sensor
307: wheel speed sensor
308: oil temperature sensor
400: internal combustion engine
410: intake passage
411: throttle valve

The invention claimed is:

1. A hydraulic pressure controller for a continuously variable transmission that changes hydraulic pressure supplied to each pulley of the continuously variable transmission to change a gear ratio, the hydraulic pressure controller comprising:

an oil pump;

a regulator valve that adjusts hydraulic pressure of hydraulic oil discharged from the oil pump and output as a line pressure by bifurcating the oil discharged from the oil pump; and

a first and a second control valve each of which further adjusts the line pressure and outputs a control pressure as hydraulic pressure supplied to each of the pulleys of the continuously variable transmission;

a hydraulic pressure circuitry configured to execute a hydraulic pressure controller feedback, wherein the hydraulic pressure controller feedback adjusts the line pressure in accordance with a level of the hydraulic pressure supplied to the pulleys, and after driving a valve body of each control valve in the valve opening direction, the hydraulic pressure circuitry executes oscillation suppression control that temporarily suppresses movement of the valve body,

the control valve uses a driving hydraulic pressure input to the control valve to drive the valve body in the valve opening direction, and

after the hydraulic pressure controller starts to drive the valve body in the valve opening direction, the oscillation suppression control further changes the driving hydraulic pressure to suppress movement of the valve body in a valve closing direction when the valve body oscillates as the output of the driving hydraulic pressure starts.

2. The hydraulic pressure controller for the continuously variable transmission according to claim **1**, wherein

the oscillation suppression control executes the further changing of the driving hydraulic pressure by further increasing the driving hydraulic pressure in accordance with a timing at which the valve body starts to move in the valve closing direction to suppress the movement of the valve body in the valve closing direction.

3. The hydraulic pressure controller for the continuously variable transmission according to claim **1**, further comprising:

an estimating circuitry configured to estimate the temperature of the hydraulic oil,

wherein the oscillation suppression control advances a timing for the further changing of the driving hydraulic pressure as the temperature of the hydraulic oil increases.

4. The hydraulic pressure controller for the continuously variable transmission according to claim **1**, wherein the oscillation suppression control advances a timing for the further changing of the driving hydraulic pressure as the driving hydraulic pressure output to drive the valve body in the valve opening direction increases.

5. The hydraulic pressure controller for the continuously variable transmission according to claim **1**, wherein

the hydraulic pressure controller feedback adjusts the line pressure in correspondence with a change in a larger one of a hydraulic pressure supplied to a first pulley, which is coupled to an internal combustion engine, and a hydraulic pressure supplied to a second pulley, which is coupled to a vehicle wheel, and

when increasing the hydraulic pressure supplied to the first pulley, the hydraulic pressure circuitry executes the oscillation suppression control when the hydraulic pressure supplied to the first pulley becomes greater than the hydraulic pressure supplied to the second pulley.

6. A hydraulic pressure controller for a continuously variable transmission that changes hydraulic pressure supplied to each pulley of the continuously variable transmission to change a gear ratio, the hydraulic pressure controller comprising:

an oil pump;

a regulator valve that adjusts hydraulic pressure of hydraulic oil discharged from the oil pump and output as a line pressure by bifurcating the oil discharged from the oil pump; and

a first and a second control valve each of which further adjusts the line pressure and outputs a control pressure as hydraulic pressure supplied to each of the pulleys of the continuously variable transmission;

a hydraulic pressure circuitry configured to execute a hydraulic pressure controller feedback, wherein the hydraulic pressure controller feedback adjusts the line pressure in accordance with a level of the hydraulic pressure supplied to the pulleys, and after driving a valve body of each control valve in the valve opening direction, the hydraulic pressure circuitry executes oscillation suppression control that temporarily suppresses movement of the valve body, wherein

the control valve includes a first pressure chamber into which a driving hydraulic pressure will flow to move the valve body in the valve opening direction, a biasing member accommodated in the first pressure chamber to bias the valve body in a valve opening direction, and a second pressure chamber arranged at an opposite side of the first pressure chamber with the valve body arranged in between, wherein the level of the driving hydraulic pressure supplied to the first pressure chamber is changed to drive the valve body, and

after starting to drive the valve body in the valve opening direction, the oscillation suppression control suppresses the movement of the valve body by prohibiting discharge of the hydraulic oil from the second pressure chamber.

7. The hydraulic pressure controller for the continuously variable transmission according to claim **6**, further comprising:

a pressure adjustment valve including a valve body moved to connect the second pressure chamber and a discharge passage when the hydraulic pressure in the second pressure chamber increases and discharge some of the hydraulic oil from the second pressure chamber through the discharge passage; and

a closing valve that can switch between a state closing the discharge passage and a state opening the discharge passage,

wherein the closing valve closes the discharge passage to prohibit the discharge of the hydraulic oil from the second pressure chamber.

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